AN ANALYTICAL STUDY OF INDUCTION IN THE OTTO CYCLE ENGINE

12T

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SYMBOLS

A	Effective area of valve in square inches
B_r	Function of P_m and P_c for mass flow
C_p	Specific heat at constant pressure
C_v	Specific heat at constant volume
C	Flow coefficient for regular flow
$\overline{C}$	Flow coefficient for flow backwards
d_i	Characteristic valve seat diameter in inches
D_r	Function of P_c and P_{co} for adiabatic compression
E'	Energy, initial
E"	Energy, final
E_r	Function of P_c and P_m for mass flow
F_r	Function of P_m and P_a for mass flow
G	Mass flow in pounds per cylinder per cycle
h	Enthalpy per unit mass
H	Heating value of the air-fuel mixture, 1240 BTU per pound equals 1160.10^4 pounds inches
k	Ratio of specific heats for air, 1.4
l	Connecting rod length, 6.75 inches
L	Valve lift in inches
n	Engine speed in revolutions per minute
P	Pressure in pounds per square inch
P_a	Atmospheric pressure, 14.7 pounds per square inch
Q	Heat added to the system in pounds times inches
r	Crank radius, 1.875 inches
R	Gas constant for air, 640 inches times pounds per degree Rankine per pound

SYMBOLS (Continued)

t Time in seconds

T Temperature in degrees Rankine

T_a Atmospheric temperature, 530 degrees Rankine

u Internal energy per unit mass

v Specific volume

V Volume in cubic inches

V_{cl} Clearance volume

$V_{c\ max}$ Maximum cylinder volume, 43 cubic inches per cylinder

V_D Displacement volume, 36.1 cubic inches per cylinder

V_{ci} Cylinder volume when inlet valve closes, 36.4 cubic inches per cylinder

w Velocity

W Work done by the system

W_p Pumping loss in pounds times inches

GREEK LETTERS

γ Specific weight

ε Compression ration, 6.25

$\Delta\eta$ Thermal efficiency increase, per cent

λ Crank radius - connecting rod - ratio, 0.278

ω Angular velocity in degrees

φ Degrees crank angle after TDC at beginning of the intake stroke

SUBSCRIPTS

c Cylinder

co Cylinder before flow in the reverse direction

SYMBOLS (Continued)

SUBSCRIPTS

e Exhaust valve

i Inlet valve

m Intake manifold, between throttle and intake valve

t Throttle valve

o,1 State with higher and lower pressure, respectively

AN ANALYTICAL STUDY OF INDUCTION IN THE OTTO CYCLE ENGINE

SUMMARY

The purpose of this thesis is to investigate analytically the flow of air into the cylinder during the intake period of a 4-stroke multicylinder Otto engine. At low load a considerable power loss occurs which may be diminished by arranging individual throttle valves for each cylinder. The pressure variation in the cylinder during the intake period for different locations of these throttle valves is calculated and compared in the air standard cycle. The differential equations that describe the pressure in the cylinder are solved in each case by means of finite differences. The calculations show that the volume between the individual throttle valves and the intake valve influences the decrease of the power loss.

The investigations are based on the construction data for a Chevrolet engine.

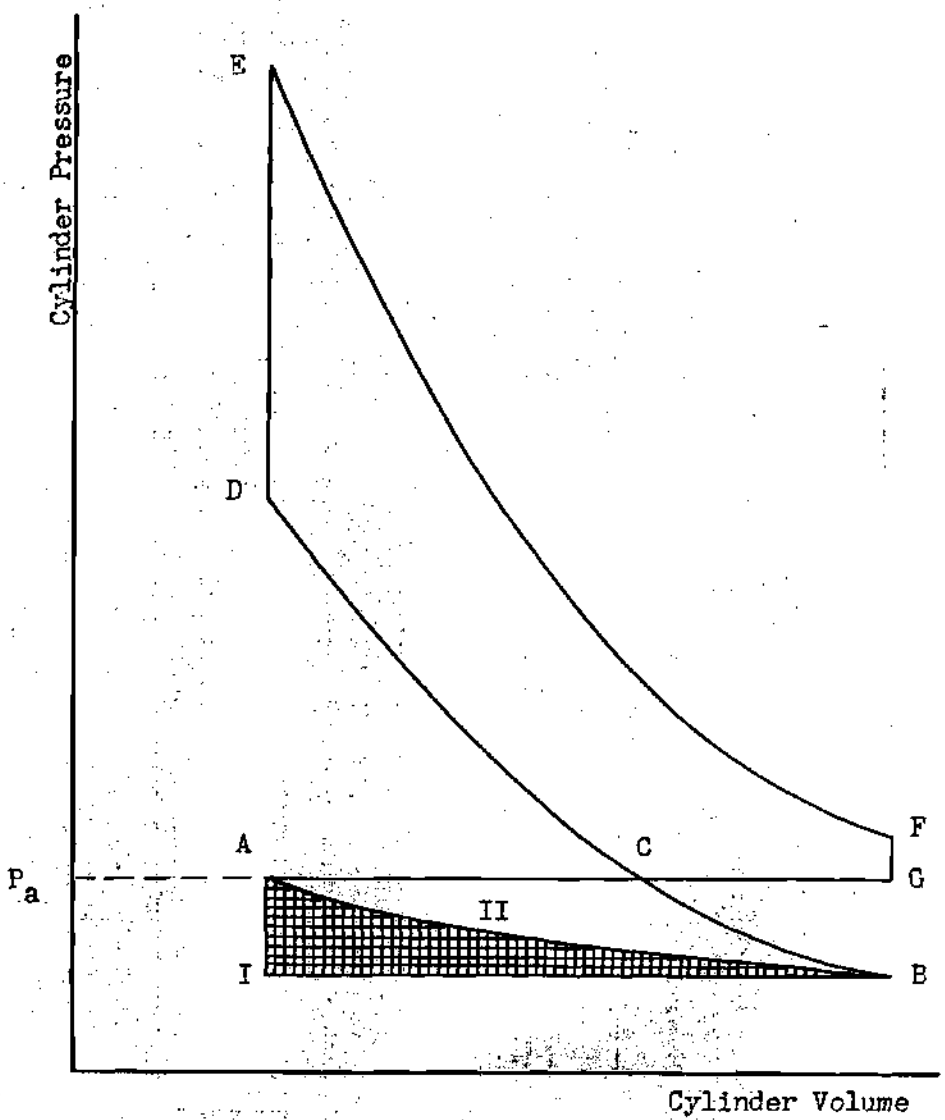
INTRODUCTION

The object of this investigation is to show by means of an analytical approach that the power loss occurring during the intake period of a 4-stroke multicylinder Otto engine can be diminished at low load by arranging separate throttle valves for each cylinder and that the choice of the location of these throttle valves is important.

The Otto cycle engine is controlled by restricting the amount of air-fuel mixture entering the cylinder during the intake stroke. The air-fuel ratio is kept constant at any load. The conventional method of accomplishing this is to use a single throttle valve in the carburetor. At a constant engine speed, the throttle valve at full load is wide open and the pressure in the intake manifold is near atmospheric. When the throttle valve is gradually closed the output of the engine decreases because of the smaller amount of mixture entering the cylinder. At the same time the pressure in the intake manifold is decreased and at very low load, this pressure can be far below atmospheric, while the pressure at the end of the exhaust stroke in the cylinder is always close to atmospheric. When the intake valve opens, a higher pressure exists in the cylinder than in the intake manifold and the relatively high-pressure residual gas expands into the intake manifold. Later, as the piston moves downward on the intake stroke, additional work has to be done in order to draw back these exhaust gases into the cylinder before the fresh charge can enter the cylinder. The entire suction stroke of the engine then occurs at the low manifold pressure and the negative work area in a pressure-volume diagram is accordingly large.

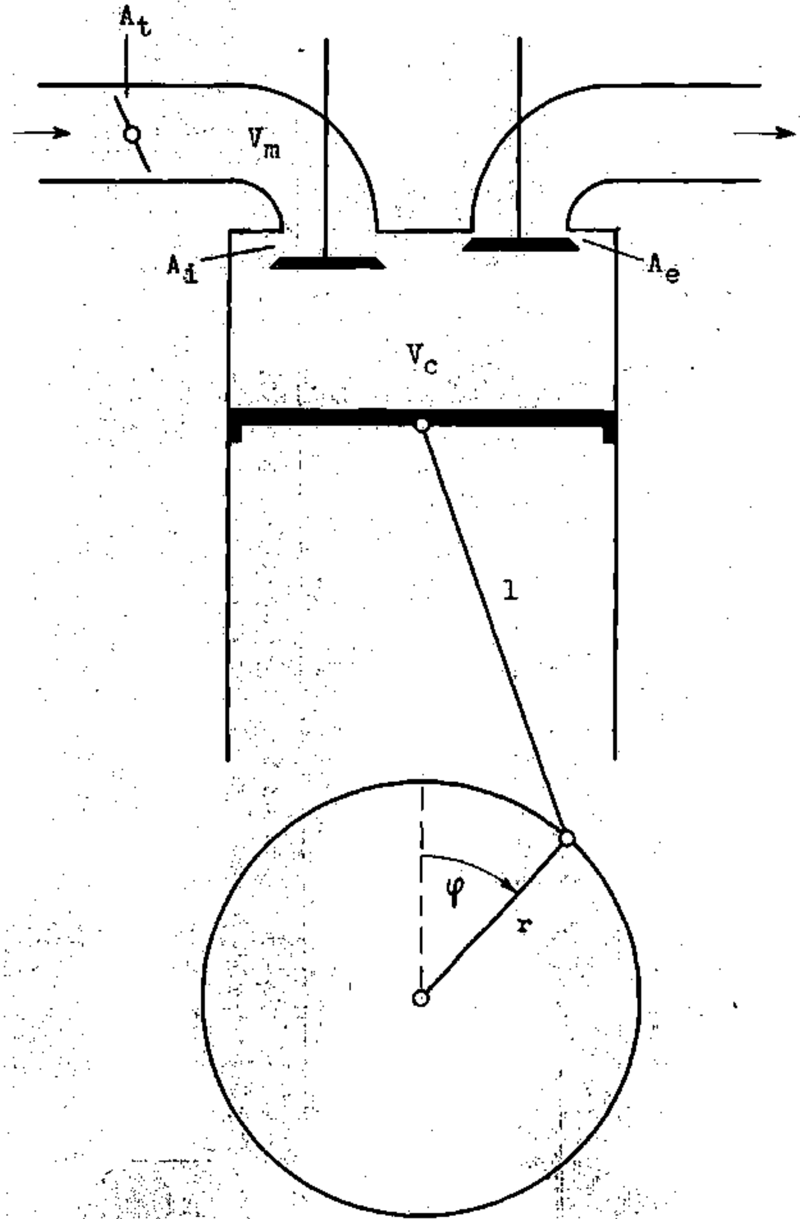
The intake process of the theoretical Otto cycle for a throttled multicylinder engine is shown by path A-I-B-C in Figure 1. In a single cylinder Otto cycle engine the conditions at low load are much better because the pressure in the manifold may build up to atmospheric pressure between the suction strokes and there is no pressure difference between cylinder and intake manifold when the intake valve opens. Consequently residual gas does not expand into the intake manifold and the pressure in the cylinder decreases gradually during the suction stroke. The corresponding process is shown by path A-II-B-C in Figure 1. Compared with the multicylinder engine there is a saving in pumping work proportional to the crosshatched area.

By using individual throttle valves for each cylinder of a multicylinder engine, similar conditions as in the single cylinder engine are obtained. It is expected that the volume (V_m) between the regular intake valve and the throttle valve controls the saving of pumping work. (See Figure 2)



Theoretical Throttled Otto Cycle

Figure 1



Schematic Drawing of a Cylinder

Figure 2

ENGINE DATA

All investigations are based on the data of a Chevrolet passenger car engine.

Engine.--The engine specifications are as follows:

Bore	inches	3.5
Stroke	inches	3.75
Displacement	cubic inches	216.5
Compression Ratio		6.25
Number of Cylinders		6
Connecting Rod length	inches	6.75

Intake Valve.--The intake valve and seat dimensions as well as the valve lift curve have been obtained from the engine in the Mechanical Engineering Laboratory. The intake valve and seat is sketched in Figure 3 and the valve lift is plotted against the degrees of crank angle in Figure 4.

Flow Coefficient for Intake Valve.--In order to get the effective flow area, the geometric flow area has to be multiplied by a coefficient which varies with the ratio valve lift to valve diameter. Information about this coefficient for the Chevrolet engine was not available.

Figure 5 shows curves for this coefficient from two different sources[1]. Both define the geometric area (A) as the cylindric area with height equal to the valve lift (L) and the diameter equal to the

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[1]Eichelberg, G., Verbrennungsmotoren, Zurich: Verlag des Schweizerischen Maschienen-Ingenieur Vereins, 1947, vo. 2, p. 24

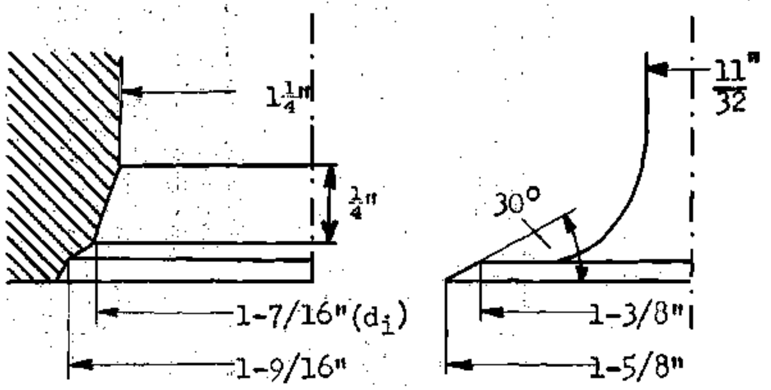
Lewis, G. W. and Nutting, E. M., Air Flow through Poppet Valves, NACA Report, no. 24, 1918, p. 65

inner seat diameter (d_i). For these investigations the coefficient given by Eichelberg are used.

Note that the flow coefficient for flow from the cylinder through the intake valve back into the intake manifold is much lower than for flow through the manifold into the cylinder. Therefore, two different effective flow areas are defined:

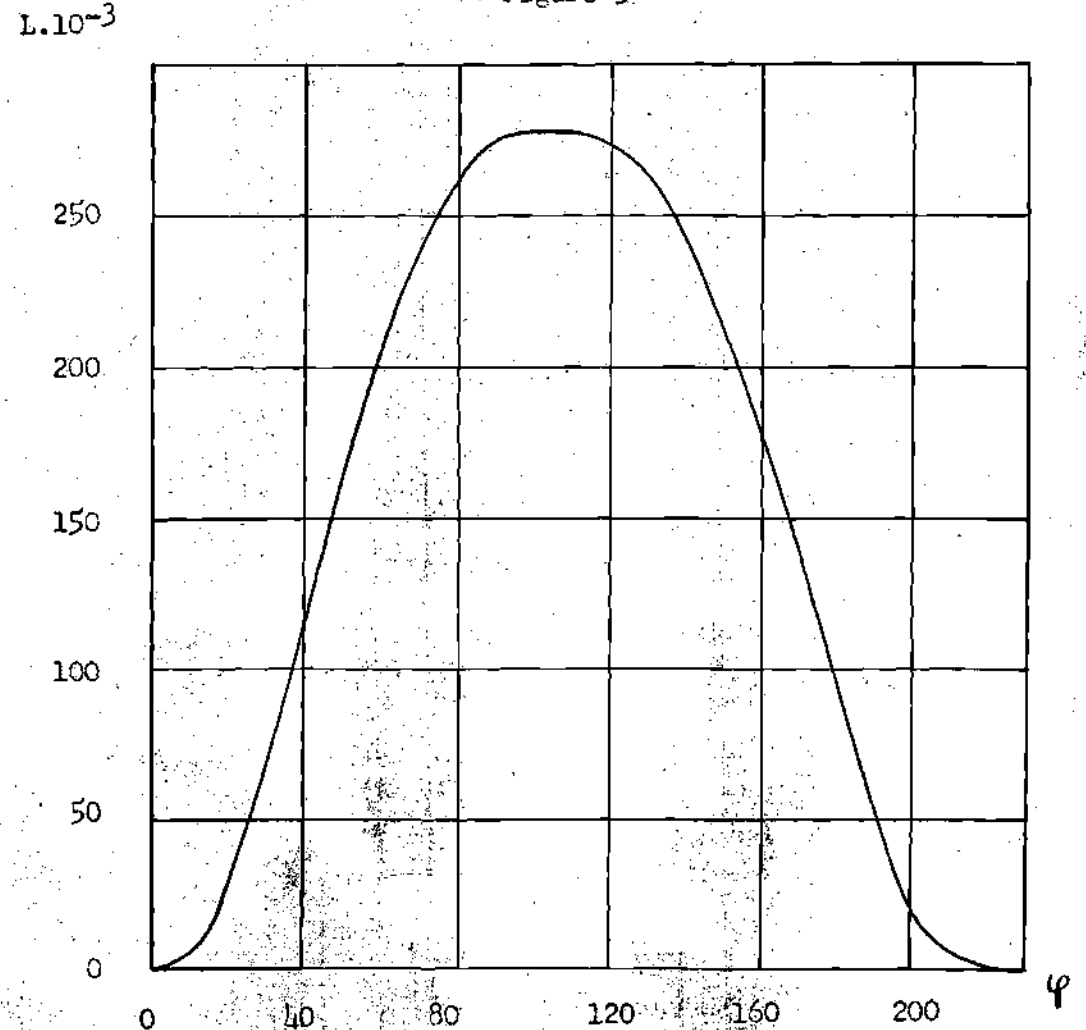
Flow into the cylinder $A_i = C.\pi.d_i.L$ (1)

Flow out of the cylinder $\bar{A}_i = \bar{C}.\pi.d_i.L$ (2)



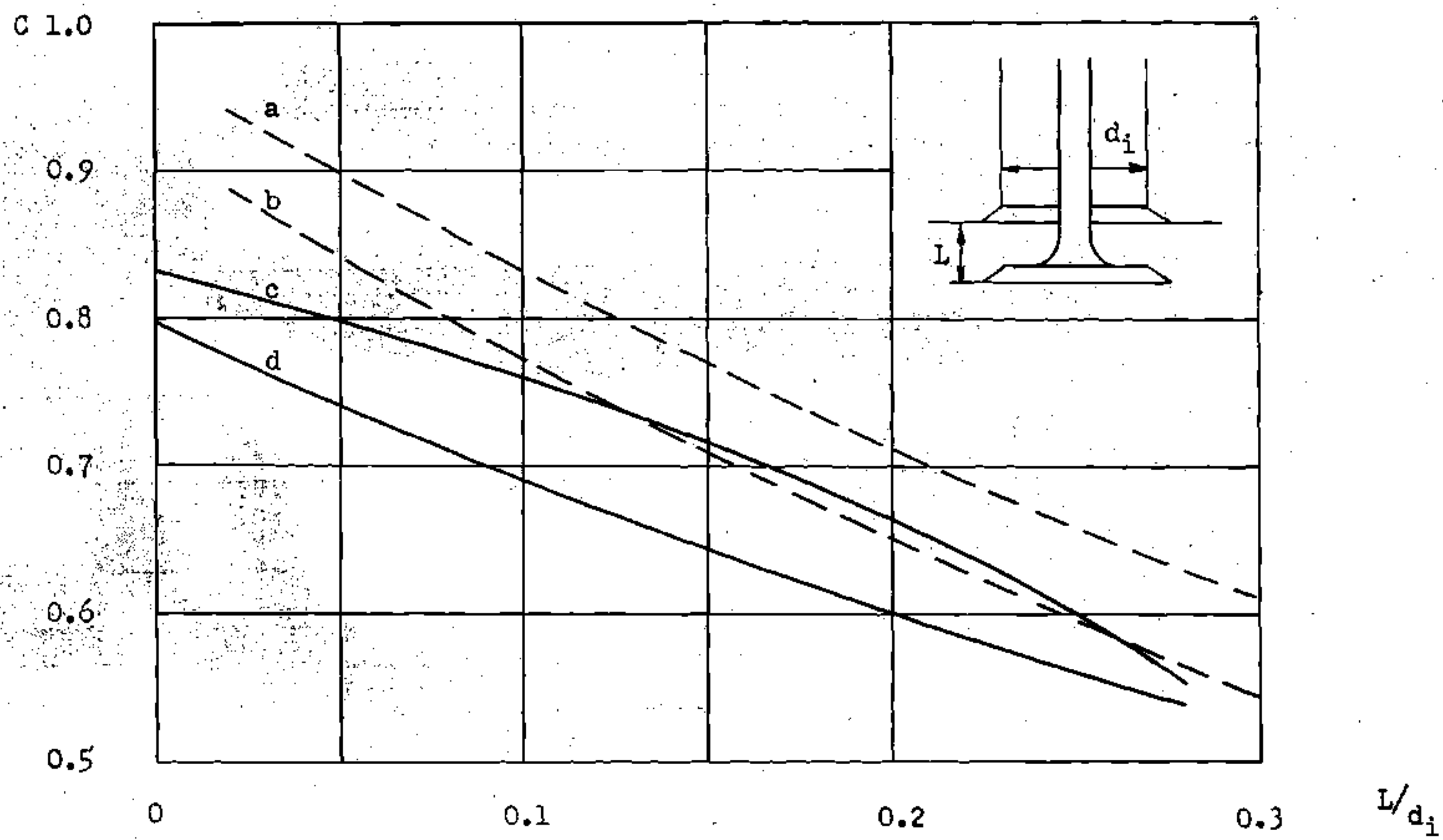
Intake Valve

Figure 3



Valve Lift Curve

Figure 4



a steady flow, two valves, $d_i = 1\text{-}3/4"$
b steady flow, two valves, $d_i = 1\text{-}1/4"$
(NACA Report)

Flow Coefficients

Figure 5

c regular flow
d flow backward
(Eichelberg, G.)

METHOD OF APPROACH

The pressure in the cylinder during the intake period is given by a system of differential equations, and the value of all variables when the intake valve opens as a boundary condition is known.

Among the various methods which could be applied to solve this system, the method of finite differences is the most convenient one.

When the pressure is found at any time during the intake period the pumping loss is obtained by graphical integration in the P-V diagram. Only the pumping loss of throttle valve arrangements which yield the same air flow to the cylinder can be compared.

In order to get a figure for the increase of the efficiency of the cycle, the saving of pumping work is compared with a fictitious total energy input during a cycle. This energy input is defined as the product of the air flow per cycle and the heating value of a normal air-fuel mixture.

The methods of finite differences:

Suppose y is an independent variable and x_1 is given by the following system of differential equations:

$$\frac{dx_1}{dy} = F_1(y, x_1, x_2, .. x_n)$$

$$\frac{dx_2}{dy} = F_2(y, x_1, x_2, .. x_n)$$

$$\frac{dx_n}{dy} = F_n(y, x_1, x_2, .. x_n)$$

If the boundary condition $y = b$, $x_1 = A_1$, $x_2 = A_2$, $x_n = A_n$ is given, an approximation for x_1 at $y' = b + \Delta y$ would be:

$$x_1' = A_1 + F_1(b, A_1, A_2, \ldots A_n) \cdot \Delta y \qquad (3)$$

By choosing Δy small enough, x_1' calculated in this way approaches the real value of x_1' as closely as desired.

When this step from y to $y' = b + \Delta y$ has been executed with all variables x, the value of x_1 at $y'' = b + 2\Delta y$ is found in the same manner. By repeating these steps simultaneously with the whole system, $x_1 = F(y)$ is found.

DERIVATION OF THE WORKING EQUATIONS

Differential Equations.--Assuming ideal gas conditions as well as instantaneous equalization of pressure variations and neglecting heat transfer from the cylinder walls to the air, it is possible to derive a system of differential equations which describe the pressure in the cylinder at any time from the conditions which are imposed by the continuity and the first law of thermodynamics. As this investigation is restricted to the intake period, it is later assumed that the exhaust valve is closed at any time.

Referring to Figures 6 and 7 and using the continuity condition for the cylinder:

$$dG_c = dG_i - dG_e$$

Substituting dG

$$d(V_c\gamma_c) = A_i w_i \gamma_i \cdot dt - A_e \cdot w_e \gamma_e \cdot dt$$

and dividing by dt

$$\frac{dV_c}{dt}\gamma_c + \frac{d\gamma_c}{dt} V_c = A_i w_i \gamma_i - A_e w_e \gamma_e$$

dividing by $\gamma_c V_c$ and rearranging yields

$$\frac{d(\ln\gamma_c)}{dt} = -\frac{d(\ln V_c)}{dt} + \frac{A_i w_i \gamma_i}{V_c\gamma_c} - \frac{A_e w_e \gamma_e}{V_c\gamma_c} \qquad (4)$$

From the P-v-T relation for ideal gas

$$\ln\gamma = \ln P - \ln R - \ln T$$

and from the relation

$$d\varphi = \omega . dt$$

$\ln\gamma$ and $d\varphi$ can be substituted in equation 4

$$\frac{d(\ln P_c)}{d\varphi} - \frac{d(\ln T_c)}{d\varphi} = - \frac{d(\ln V_c)}{d\varphi} + A_i w_i \gamma_i \frac{RT_c}{P_c V_c \omega} - A_e w_e \gamma_e \frac{RT_c}{P_c V_c \omega} \qquad (5)$$

The application of the first law of thermodynamics:

$$Q - W = E'' - E' \qquad (6)$$

The total energy of the system, before dG_i enters the cylinder and dG_e left the cylinder (See Figure 6)

$$E' = dG_i u_m + (G_c - dG_e) u_c$$

The total energy of the system after dG_i entered and dG_e left the cylinder is (See Figure 7)

$$E'' = (dG_i + G_c - dG_e)(u_c + du_c)$$

The kinetic energy in both terms is neglected. There is no heat added to the system and $Q = 0$. The total work done by the system during this process is

$$W = dG_e v_c P_c + P_c dV_c - dG_i v_m P_m$$

Substituting the terms Q, W, E', E" in equation 6 and neglecting terms of second order leads to

$$C_v dG_i (kT_m - T_c) = C_v \frac{P_c V_c}{RT_c} dT_c + P_c dV_c + (C_p - C_v) T_c dG_e$$

when dG_i and dG_e are replaced by

$$dG_i = A_i w_i \gamma_i \frac{d\varphi}{\omega}$$
$$dG_e = A_e w_e \gamma_e \frac{d\varphi}{\omega}$$

and the above equation is rearranged, equation 7 is obtained

$$\frac{d(\ln T_c)}{d\varphi} = -(k-1)\frac{d(\ln V_c)}{d\varphi} + A_i w_i \gamma_i \frac{R(kT_m - T_c)}{\omega . P_c V_c} - A_e w_e \gamma_e \frac{R(k-1)T_c}{\omega . P_c V_c} \quad (7)$$

Subtracting equation 7 from equation 5 yields

$$\frac{d(\ln P_c)}{d\varphi} = -k\frac{d(\ln V_c)}{d\varphi} + kA_i w_i \gamma_i \frac{RT_m}{\omega P_c V_c} - kA_e w_e \gamma_e \frac{RT_c}{\omega P_c V_c} \quad (8)$$

Applying the same procedure, which led to equations 7 and 8 on the volume (V_m) between the throttle valve and the intake valve instead of the cylinder, two corresponding equations for P_m and T_m are obtained.

$$\frac{d(\ln T_m)}{d\varphi} = w_t A_t \gamma_t \frac{R(kT_a - T_m)}{\omega P_m V_m} - A_i \gamma_i w_i \frac{R(k-1)T_m}{\omega P_m V_m} \quad (9)$$

$$\frac{d(\ln P_m)}{d\varphi} = w_t A_t \gamma_t \frac{RT_a \cdot k}{\omega P_m V_m} - A_i \gamma_i w_i \frac{kRT_m}{\omega P_m V_m} \quad (10)$$

The volume, V_m, is constant, and the term $Pk\frac{d(\ln V)}{dP}$ does not appear.

The next step is to express $w \gamma$ in the last three equations, nos. 8, 9, and 10, by the temperature and pressure on both sides of the corresponding valve. Suppose P_o and P_1 are these pressures and $P_o > P_1$, $w\gamma$ yields[2] for adiabatic expansion

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[2]Keenan, J. H. and Kaye, J., Gas Tables, New York: John Wiley and Son, Inc., 1948, p. 130.

$$\gamma \cdot w=\sqrt{\frac{2g}{R}\cdot\frac{k}{k-1}}\frac{P_o}{\sqrt{T_o}}\sqrt{\left(\frac{P_1}{P_o}\right)^{\frac{2}{k}}-\left(\frac{P_1}{P_o}\right)^{\frac{k+1}{k}}}=\sqrt{\frac{2g}{R}\cdot\frac{k}{k-1}}\frac{P_o}{\sqrt{T_o}}\cdot F\left(\frac{P_1}{P_o}\right) \tag{11}$$

For $F(\frac{P_c}{P_m})$, $F(\frac{P_m}{P_a})$, $F(\frac{P_m}{P_c})$ the symbols E_r, F_r, and B_r are used. Values for this kind of function are contained in the Gas Tables by the authors just cited. These symbols are now introduced into the equations 8, 9, 10, and d(lnP), d(lnT) on the left side of these equations are expanded.

$$\frac{dP_c}{d\varphi}=-P_c k\frac{d(\ln V_c)}{d\varphi}+\frac{k.R}{\omega}\sqrt{\frac{2g}{R}\;\frac{k}{k-1}}\;\frac{A_i}{V_c}\sqrt{T_m}P_m E_r \tag{12}$$

$$\frac{dT_m}{d\varphi}=\frac{R}{\omega}\cdot\frac{A_t}{V_m}\cdot\frac{P_a}{\sqrt{T_a}}\sqrt{\frac{2g}{R}\;\frac{k}{k-1}}\;\frac{T_m}{P_m}(kT_a-T_m)F_r-\frac{R}{\omega}\sqrt{\frac{2g}{R}\cdot\frac{k}{k-1}}\;\frac{(k-1)}{V_m}A_i\sqrt{T_m}^3 E_r \tag{13}$$

$$\frac{dP_m}{d\varphi}=\frac{kR}{\omega}\;\frac{A_t}{V_m}\sqrt{T_a}P_a\sqrt{\frac{2g}{R}\;\frac{k}{k-1}}\cdot F_r-\frac{kR}{\omega}\sqrt{\frac{2g}{R}\;\frac{k}{k-1}}\;\frac{1}{V_m}\;A_i\sqrt{T_m}\cdot P_m E_r \tag{14}$$

When the air flows in the reverse direction, i.e., from the cylinder into the intake manifold, these equations have to be modified. In this case, only $\frac{dP_c}{d\varphi}$ is required for this investigation.

$$\frac{dP_c}{d\varphi}=-P_c k\frac{d(\ln V_c)}{d\varphi}-\frac{kR}{\omega}\sqrt{\frac{2g}{R}\;\frac{k}{k-1}}\;\frac{\bar{A}_i}{V_c}\sqrt{T_c}P_c B_r \tag{15}$$

The temperature in the cylinder follows the rule of adiabatic expansion

$$T_c=T_{c_o}\left(\frac{P_c}{P_{c_o}}\right)^{\frac{k-1}{k}}=T_{c_o}\cdot D_r \tag{16}$$

where the subscript c_o indicates the state at the beginning of flow in the reverse direction.

The flow through the valve is a throttling process and the temperature of this gas when it arrives in the intake manifold is equal to T_c at any time. It is assumed that there is no mixing with the fresh air in the manifold. The equations 12, 13, and 14 represent a system of differential equations which is to be solved for P_c as indicated in the previous chapter.

The Variation of V_c with φ.—In equation 12 the terms V_c and $\frac{d(\ln V_c)}{d\varphi}$ appear and the values of these terms as a function of φ measured in degrees from the TDC are required. These values are determined by the compression ratio($\mathcal{E}$), the crank radius - connecting rod-ratio (λ) and the displacement volume (V_D) of one cylinder.

$$V_c = V_{cl} + V_x = V_D\left[\frac{V_{cl}}{V_D} + \frac{V_x}{V_D}\right] = V_D\left[\frac{1}{\mathcal{E}-1} + \frac{V_x}{V_D}\right] \quad (17)$$

V_{cl} denotes the clearance volume and V_x is equal to V_c-V_{cl}

$$\frac{V_x}{V_D} = \frac{1}{2}(1 - \cos\varphi \pm \frac{\lambda}{2} \sin^2\varphi \ ) \quad (18)^3$$

The negative sign is always used for decreasing V_c.

Substituting equation 18 in equation 17

$$V_c = \tfrac{1}{2}V_D\left[\frac{2}{\mathcal{E}-1} + 1 - \cos\varphi \pm \frac{\lambda}{2} \sin^2\varphi\right] \quad (19)$$

From this equation V_c can be found for every value of φ.

$$\frac{dV_c}{d\varphi} = \frac{dV_x}{d\varphi} = V_D(\sin\varphi \pm \frac{\lambda}{2} \sin 2\varphi)\frac{\pi}{360} \quad 3$$

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[3]Dubel, H. Taschenbuch fuer den Maschienenbau, Berlin, Gottingen, Heidelberg: Springer Verlag, 1949, vo. 1, p. 598.

$$\frac{d(\ln V_c)}{d\varphi} = \frac{dV_c}{V_c \cdot d\varphi} = \frac{V_D}{V_c}(\sin\varphi \pm \frac{\lambda}{2}\sin 2\varphi)\frac{\pi}{360}$$

Substituting V_c from equation 19

$$\frac{d(\ln V_c)}{d\varphi} = \frac{(\sin\varphi \pm \frac{\lambda}{2}\sin 2\varphi)\frac{2\pi}{360}}{\frac{2}{\varepsilon - 1} + (1 - \cos\varphi \pm \frac{\lambda}{2}\sin^2\varphi)} \tag{20}$$

The values of V_c and $\frac{kd(\ln V_c)}{d\varphi}$ in function of φ are indicated in Table I. Other terms which appear in the differential equations and which are determined by the engine specifications and φ are contained in the same table. These terms are $A_i, \frac{A_i}{V_c}, \left(\frac{V_{c_i}}{V_c}\right)^k$.

The last term is used for calculation of P_c after the inlet valve has closed.

$$P_c = P_{c_i}\left(\frac{V_{c_i}}{V_c}\right)^k \tag{21}$$

The subscript c_i indicates the state when the inlet valve closes.

Mass Flow.--The mass dG flowing through an opening with the area A during the time dt is

$$dG = Aw\gamma \cdot \frac{d\varphi}{\omega}$$

and using equation 11

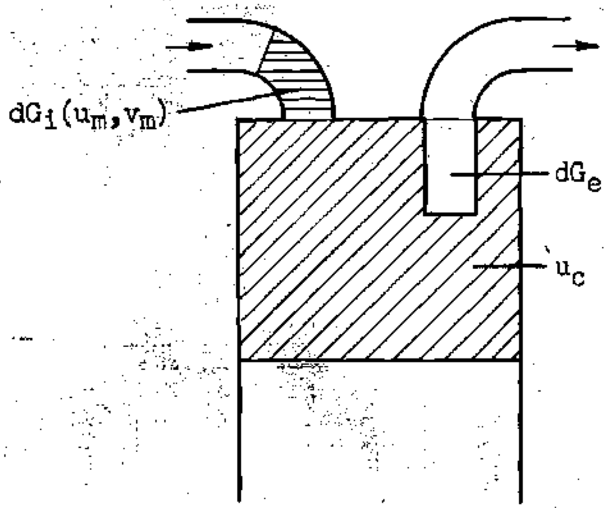
$$dG = A \cdot \sqrt{\frac{2g}{R}\,\frac{k}{k-1}}\;\frac{P_o}{\sqrt{T_o}}\;F\left(\frac{P_1}{P_o}\right) \cdot \frac{d\varphi}{\omega}$$

Integrated for a cycle:

$$G = \frac{1}{\omega}\sqrt{\frac{2g}{R}\,\frac{k}{k-1}}\int_{0}^{720}\frac{P_o}{\sqrt{T_o}}\,.A.F\left(\frac{P_1}{P_o}\right)\,d\varphi \qquad (22)$$

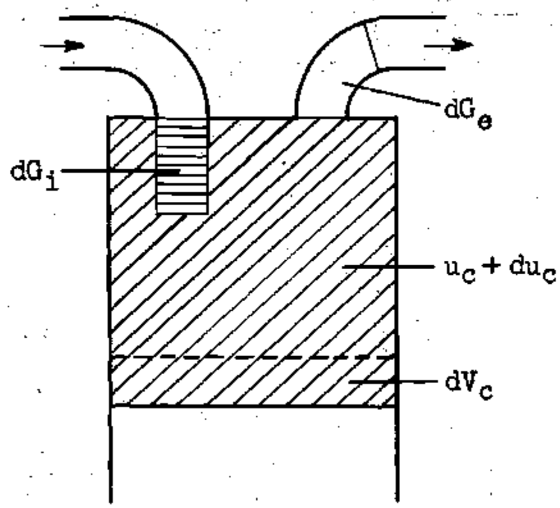
This equation has to be applied to either the throttle valve or the intake valve. Subscript o indicates again the state with the higher pressure.

(The "System" is indicated by the crosshatched area)



Mass and Energy at Time t

Figure 6



Mass and Energy at Time t + dt

Figure 7

DISCUSSION

The product G.n is an indication of the power input, where G denotes the air flow and n the engine speed. Investigating the cycle in the P-V diagram, G is an indication of the energy input per cycle and n is a parameter.

<u>The Influence of A_t and n on G and P_c.</u>--The case of the fully opened throttle valve is first calculated in order to get some information about the maximum air flow into the cylinder. Among the equations 12, 13, and 14 only equation 12 has been used[1] for the pressure calculation because T_m and P_m are equal to T_a and P_a at any time. The area enclosed by the pressure line for the intake period and by the pressure line for the exhaust period (the latter is assumed to be the atmospheric pressure line), plotted versus V_c instead of versus φ; represents the pumping loss. The curves in Figure 8 show that atmospheric pressure is built up in the cylinder immediately before the intake valve closes. The air flow is obtained by integrating graphically equation 22 and is $15.06 \cdot 10^{-4}$, and $14.36 \cdot 10^{-4}$ pounds per cycle at 2000 and 3000 rpm respectively.

In order to get some quantitative information about the effect of throttling on the air flow and the pumping loss, an individual

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[1]The pressure is calculated in intervals of 12.5 degrees of crank angle in general. When the slope of the P-φ curve changes much with increasing φ, it was necessary to diminish these intervals.

throttle valve was located in front of the air intake valve. The flow area of the throttle valve is 0.08 square inches and the volume between the latter and the air intake valve is zero. P_c and G are calculated in the same way as above, considering that the air flow is controlled by the constant throttle valve opening, when the latter is smaller than the intake valve opening.

The corresponding curves in Figure 9 show that the pumping losses are higher than in the previous case. According to the lower air-flow $8.76 \cdot 10^{-4}$; $4.38 \cdot 10^{-4}$; $2.92 \cdot 10^{-4}$ pounds per cycle at 1000, 2000, 3000 rpm respectively the pressure in the cylinder no longer reaches atmospheric pressure at the intake valve closing. The pressure after intake valve closing was found by means of equation 21.

The Influence of V_m on the Pumping Loss.—The effect of different locations of the throttle valve can be compared only when G and n have the same value. Therefore, different throttle valve arrangements at $n = 3000$ rpm, all of which result in the same air flow, are investigated below.

In the first arrangement, V_m is again zero and the throttle valve opening is 0.123 square inches. The corresponding P_c curve is plotted in Figures 10 and 13 (A-I-B-C), and the air flow is 4.16 pounds per cycle.

In the second case, V_m is made large enough so that the pressure P_m remains constant at any time. In order to get again the same air flow of 4.16 pounds per cycle as above, P_m was found to be 5.2 pounds per square inch. This represents the conventional arrangement with one throttle valve for all cylinders. At intake valve opening, exhaust gases flow into the intake manifold until $\varphi=34$ degrees after top dead center and are forced back into the cylinder. At $\varphi=59$ degrees after top dead center the first particle of the fresh charge enters the cylinder (Figures 11 and 13, curve A-II-B'-C). For this calculation it was assumed that the exhaust gases do not mix with the fresh charge in the intake manifold and that the gas temperature in the cylinder at intake valve opening is 2300 degrees Rankine.

A comparison of the pumping loss of this case with the pumping loss in the previous case shows that the latter is only 15.85 per cent smaller. The individual throttle valve does not greatly affect the pumping loss when $V_m = 0$.

In a third arrangement V_m is chosen so that there is just time enough left between two suction strokes to build up atmospheric pressure in this chamber. V_m was found to be one-third of the cylinder volume and A_t equals 0.04 square inches for the conditions $n = 3000$ rpm, $G = 4.16$ pounds per cycle. In this case equations 12 and 14 have been used. T_m, having only little influence on P_c, was assumed to follow P_m as in an adiabatic expansion. A previous calculation using equation 13 showed that this simplification is acceptable.

The pressure in the intake manifold and in the cylinder is plotted in Figure 12 versus φ and the pumping loss is found by graphical integration of curve A-III-B-C in Figure 13.

A comparison with the pumping loss for the conventional valve arrangement shows that the pumping loss A-II-B'-C is 50.5 per cent smaller than the latter.

When this saving of pumping work is compared with a fictitious energy input G.H, where H denotes the heating value of the air-fuel mixture (H=1240 BTU per pound), it corresponds to an increase in thermal efficiency of the cycle of an additional 1.72 per cent. Suppose the thermal efficiency at this low load is 10 per cent, then the improved thermal efficiency would be 11.72 per cent.

It will be recalled that this favorable V_m varies with the engine load and speed. For the choice of V_m in an actual engine, the operating conditions have to be considered. In any condition, V_m will be bigger than it would be for the lowest energy input and highest speed. Moreover, this highly satisfactory result is obtained, assuming that this change in the conventional arrangement does not affect vaporization of the fuel and combustion as it happened in the experiments of E. J. Shatz[4]. It is therefore suggested to use individual carburetors or to replace the carburetion system by fuel injection into the cylinder.

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[4]Schatz, E. J., An Investigation of the Effects of the Use of Individual Throttle Valves for Each Cylinder of a Multi-Cylinder Internal Combustion Engine. Unpublished M. S. Thesis, Georgia Institute of Technology, 1952.

RESULTS

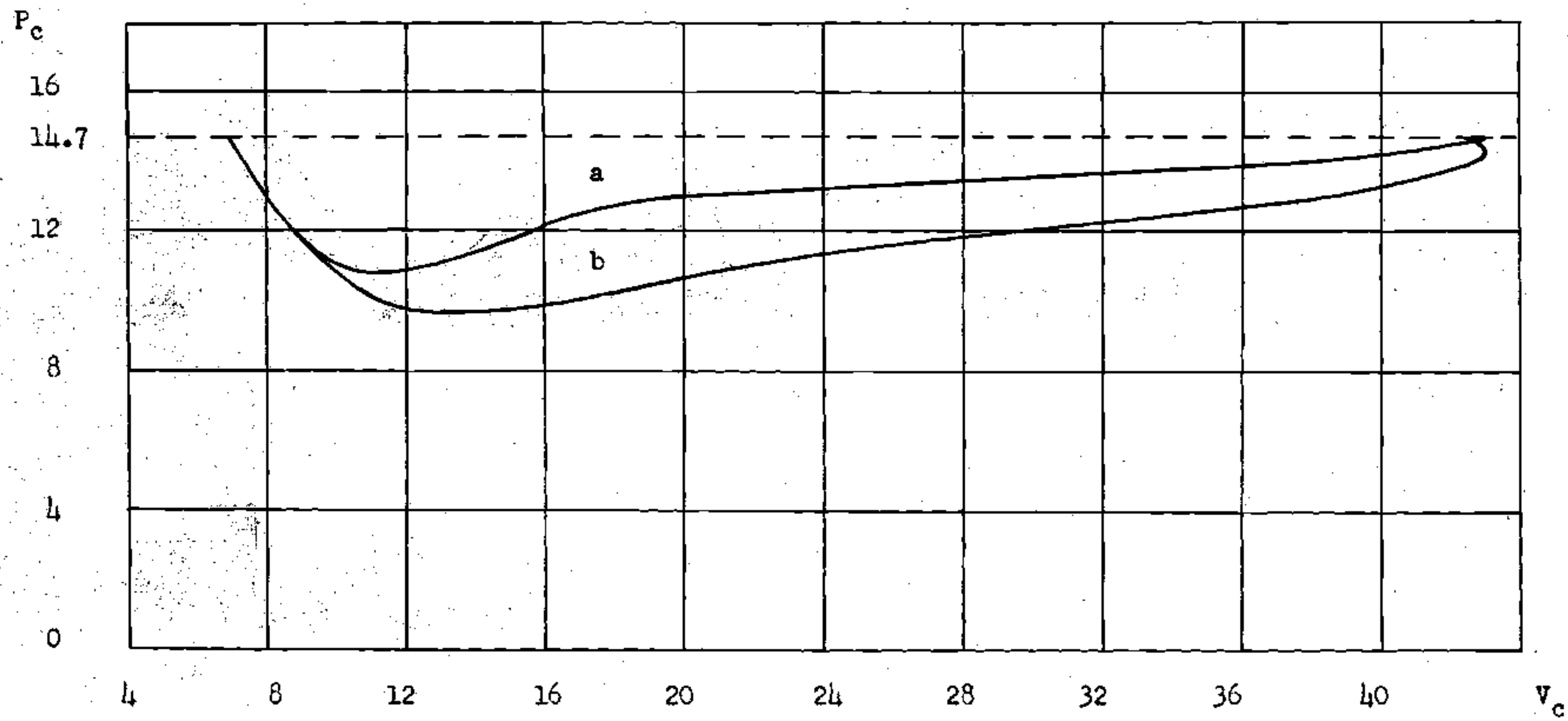
1. This investigation of the air standard cycle of the Otto cycle engine shows that an highly satisfactory improvement in thermal efficiency is possible at part load, by using separate throttle valves.

2. The volume between the individual throttle valve and the intake valve is of great importance. An arrangement with the throttle valve immediately in front of the intake valve does not result in appreciable improvement. It seems that the best conditions are reached when the volume between the throttle valve and intake valve is such that there is just time enough left to build up atmospheric pressure in this chamber between two suction strokes. This volume was one-third of the maximum cylinder volume in the case investigated and depends on engine speed and load.

APPENDIX

Table I. Variables Which Depend Only on φ

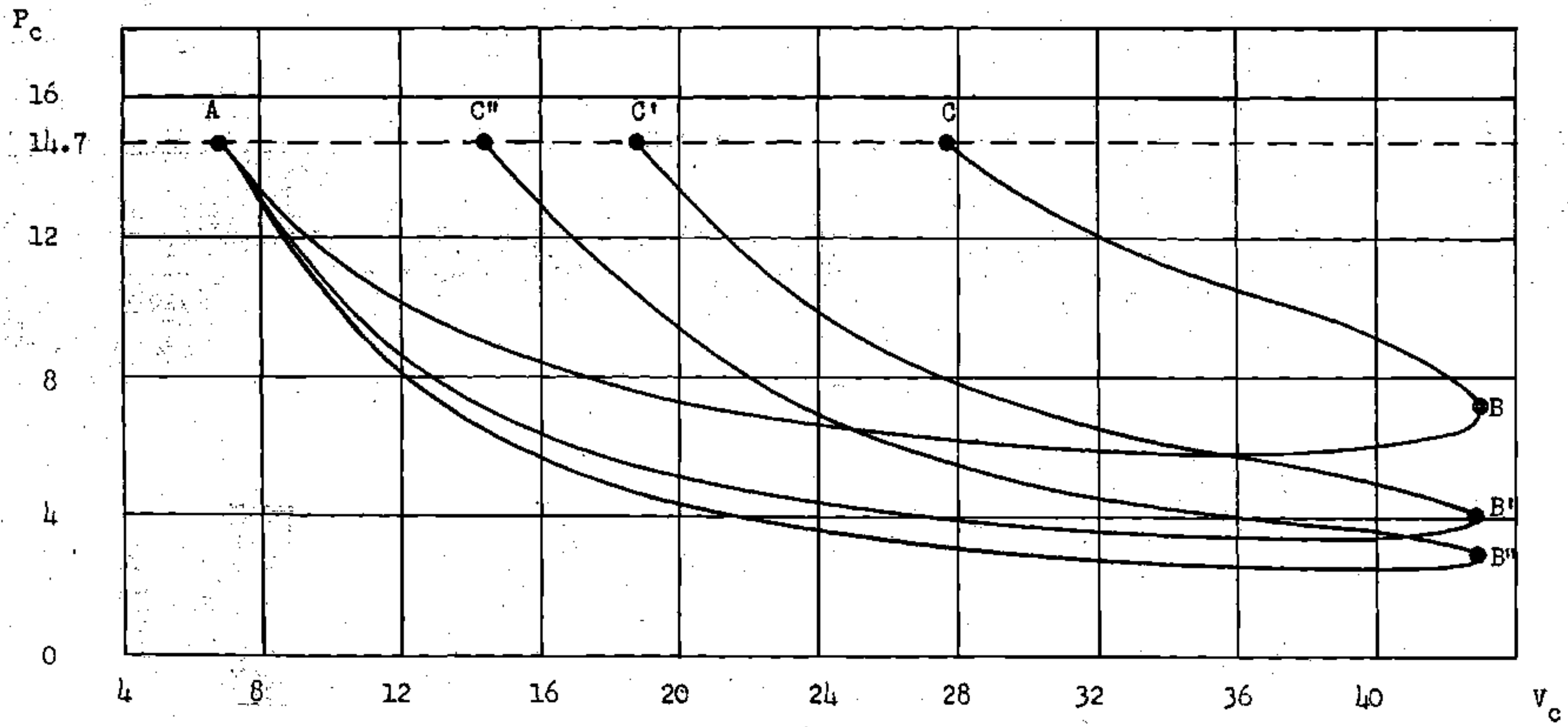
φ	V_c	$k\frac{d(\ln V_c)}{d\varphi}$		A_i	$\frac{A_i}{V_c}$		$\left[\frac{V_{c_i}}{V_c}\right]^k$
-9	7.03	-0.711	10^{-2}	0.000	0.000 .	10^{-2}	
0	6.88	0.000	"	0.006	0.082	"	
25.5	7.42	1.635	"	0.050	0.679	"	
25	9.01	2.590	"	0.186	2.070	"	
37.5	11.53	2.840	"	0.363	3.140	"	
50	14.80	2.690	"	0.549	3.710	"	
62.5	18.55	2.375	"	0.686	3.700	"	
75	22.61	2.020	"	0.778	3.440	"	
87.5	26.61	1.675	"	0.837	3.140	"	
100	30.50	1.355	"	0.841	2.760	"	
112.5	34.00	1.075	"	0.841	2.475	"	
125	36.95	0.820	"	0.826	2.235	"	
137.5	39.40	0.600	"	0.778	1.975	"	
150	41.23	0.405	"	0.695	1.685	"	
167.5	42.35	0.229	"	0.579	1.365	"	
175	42.90	0.065	"	0.408	0.953	"	
180	43.00	0.000	"	0.333	0.775	"	
187.5	42.75	-0.172	"	0.221	0.517	"	
200	41.60	-0.456	"	0.074	0.178	"	
212.5	39.45	-0.740	"	0.011	0.029	"	
225	36.40	-1.050	"	0.000	0.000	"	1
237.5	32.80						1.16
250	28.90						1.31
262.5	24.85						1.71
275	20.90						2.18
300	14.00						3.81
325	9.32						6.76



Pumping Loss, Throttle Valve Fully Opened

a: $n = 2000$ rpm $G = 15.06.10^{-4}$
b: $n = 3000$ rpm $G = 14.36.10^{-4}$

Figure 8



Pumping Loss of the Throttled Engine: $V_m = 0$, $A_t = 0.08$
A-B-C : $n = 1000$ $G = 8.76 . 10^{-4}$
A-B'-C' : $n = 2000$ $G = 4.38 . 10^{-4}$
A-B"-C" : $n = 3000$ $G = 2.92 . 10^{-4}$

Figure 9

Figure 10

Cylinder and Manifold Press.
n = 3000 $G = 4.16.10^{-4}$
$V_m = 0$ $A_t = 0.123$

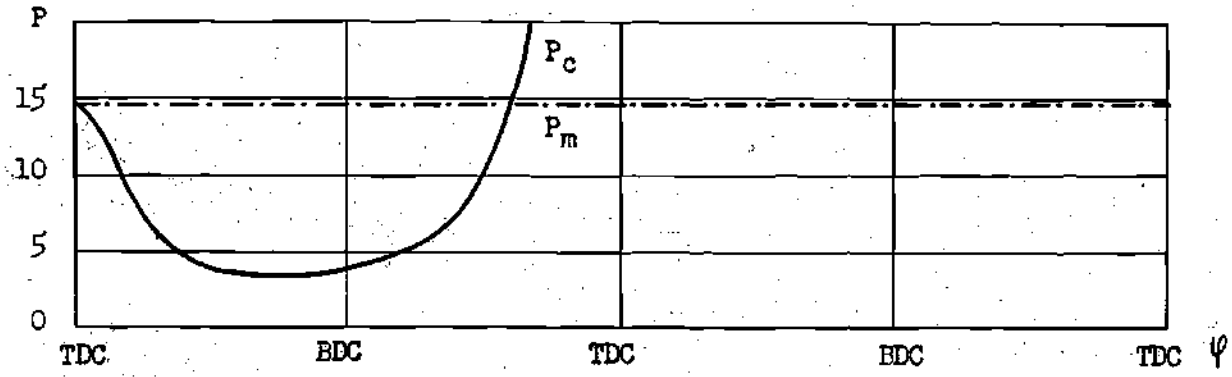


Figure 11

Cylinder and Manifold Press.
n = 3000 $G = 4.16.10^{-4}$
$V_m = \infty$ $P_m = 5.2$

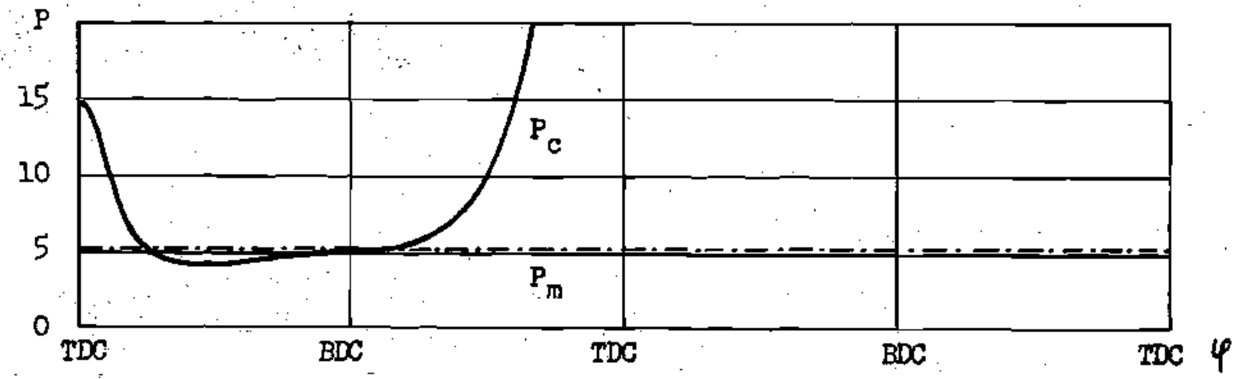
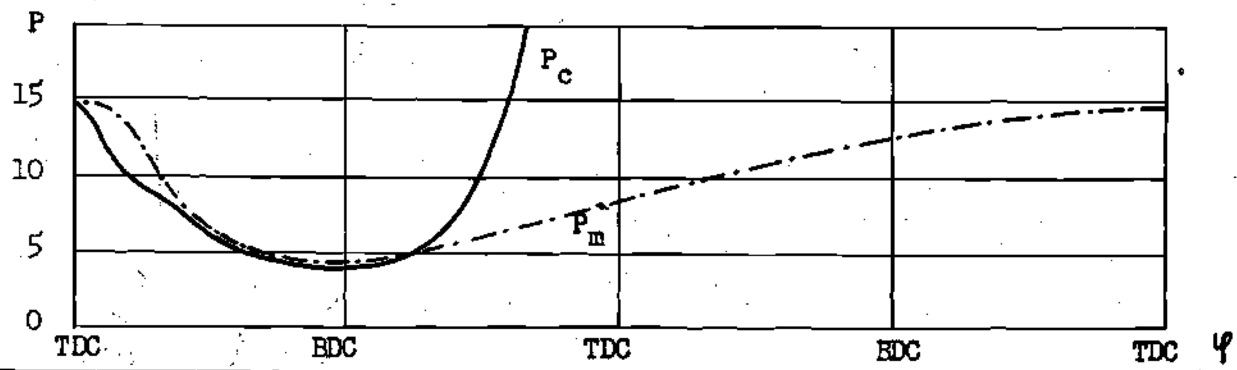
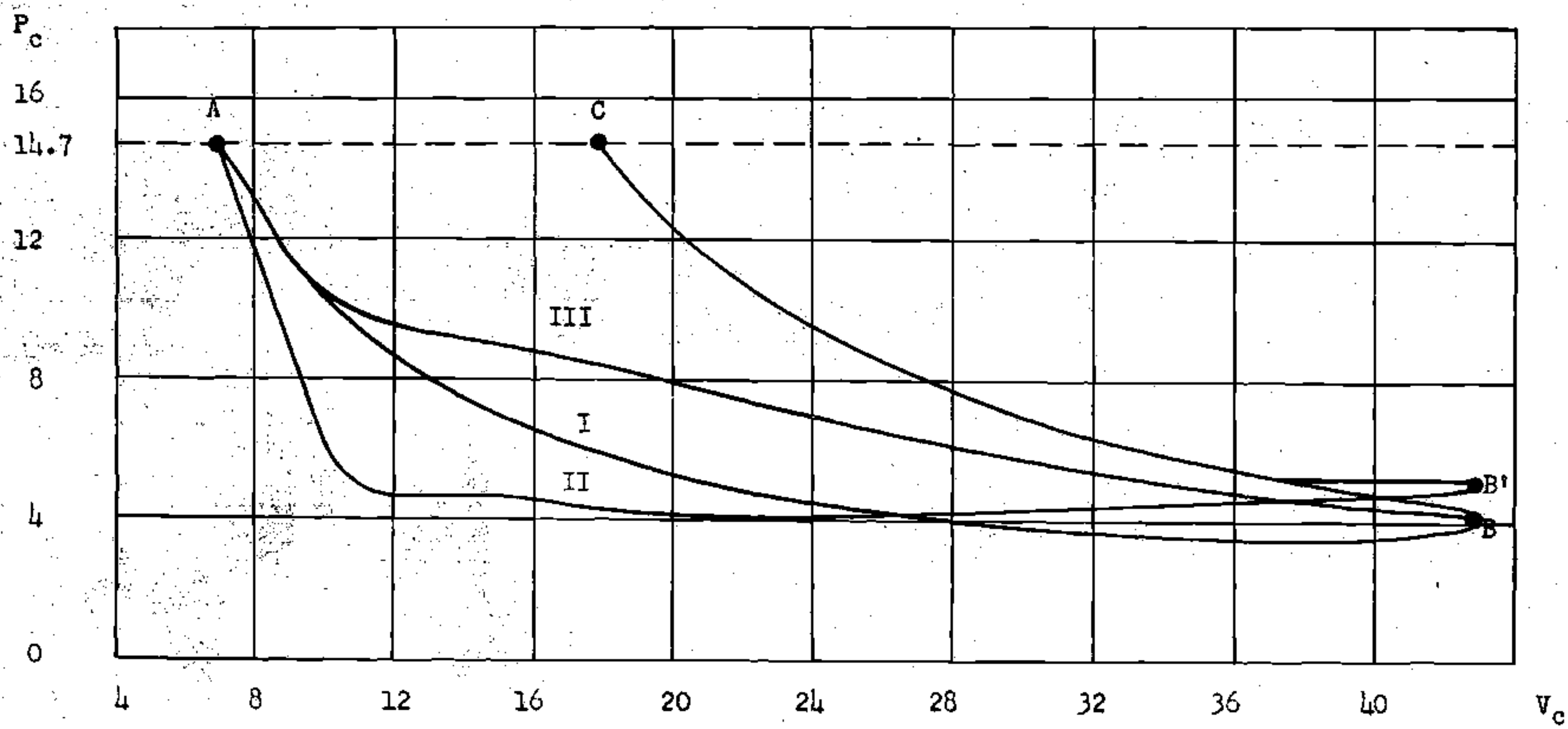


Figure 12

Cylinder and Manifold Press.
n = 3000 $G = 4.16.10^{-4}$
$V_m = 14.33$ $A_t = 0.04$





Pumping Loss of the Throttled Engine: n 3000 G $4.16.10^{-4}$

A-I-B-C	:	$V_m = 0$	$A_t = 0.123$	W_p I =138
A-II-B'-C	:	$V_m = \infty$	$P_m = 5.2$	W_p II=164
A-III-B-C	:	$V_m = 14.33$	$A_t = 0.04$	W_p III=81

Figure 13

SAMPLE CALCULATION

Air flow for $n = 3000$, $V_m = 14.33$, $A_t = 0.04$

$$G = \frac{1}{\omega}\sqrt{\frac{2g.k}{R(k-1)}} \cdot \frac{P_a}{\sqrt{T_a}} A_t \cdot \int_0^{720} F_r . d\varphi \qquad \text{(Equation 23)}$$

$$\omega = n.\frac{360}{60} = 18000$$

$$\int_0^{720} F_r . d\varphi = \int_0^{720} \sqrt{\left[\frac{P_m}{P_a}\right]^{\frac{2}{k}} - \left[\frac{P_m}{P_a}\right]^{\frac{k+1}{k}}} . d\varphi = 14.06 \qquad \text{(See Figure 14)}$$

$$G = \frac{2.05 \times 14.7 \times 0.04 \times 14.06}{18000 \times \sqrt{530}} = \underline{4.16.10^{-4}}$$

Thermal efficiency increase for $n = 3000$, $G = 4.16.10^{-4}$,

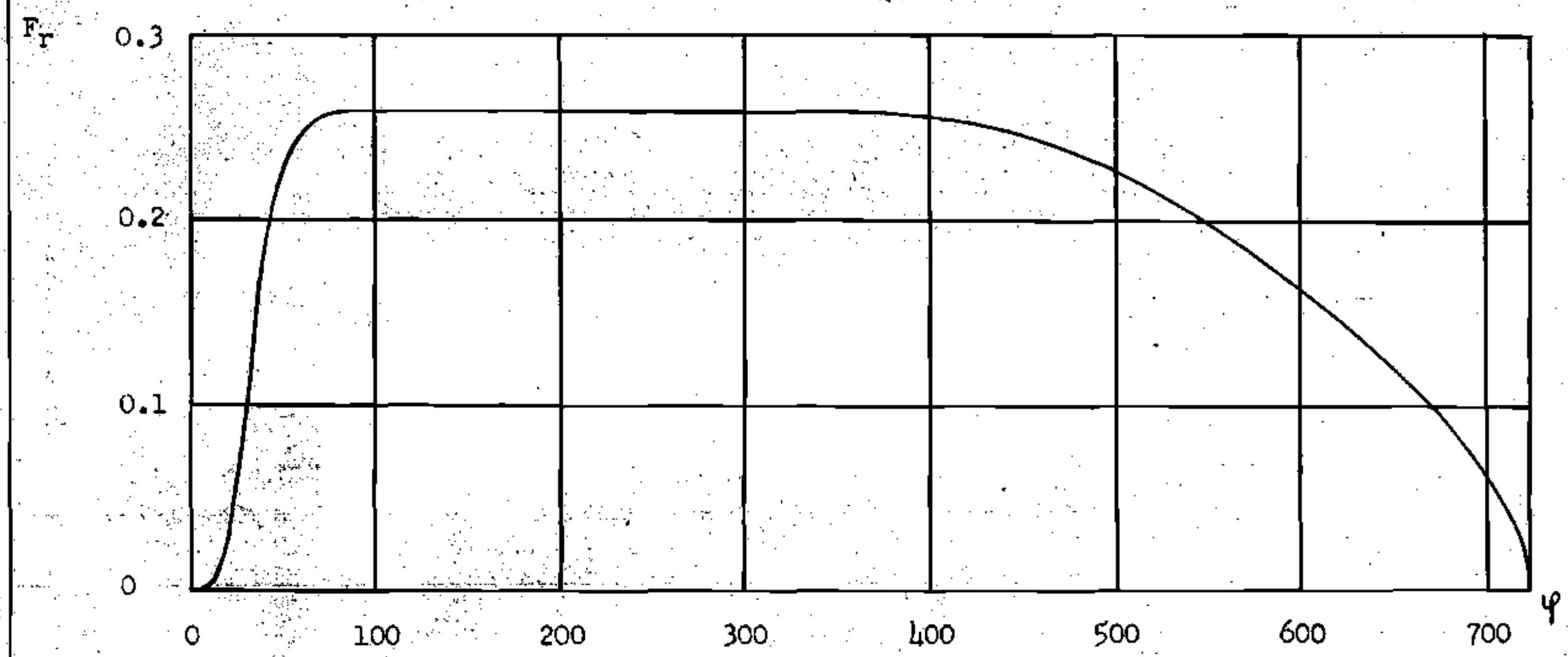
$V_m = \frac{1}{3} V_{c\ max}$

$$\Delta\eta = \frac{\Delta W_p}{Q} . 100$$

$$\Delta W_p = W_{p\ II} - W_{p\ III} = 164 - 81 = 83 \qquad \text{(See Figure 13)}$$

$$Q = G.H \quad 4.16.10^{-4} \times 1160.10^{-4} = 4830$$

$$\Delta\eta = \frac{83}{4830} \times 100 = \underline{1.72}$$



Curve for Flow Calculation

$$\int_0^{720} F_r . d\varphi = 14.06$$

Figure 14

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